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Extensive thermodynamic and economic analysis of the cogeneration of heat and power system fueled by the blend of natural gas and biogas

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ABSTRACT

Due to the finite resources of fossil fuels and their role as the most important factor in air pollution and climate change, the use of renewable energy sources is the best solution to meet the power requirements. One of the appropriate technologies in terms of energy saving and power generation with high efficiency is using the cogeneration of heat and power in which renewable fuels are used instead of fossil fuels. In the present work, a cogeneration system, consisting of a gas turbine cycle with power generation capacity of 30 MW, a steam generator, an anaerobic digester, and a heat exchanger for heating the digester and pretreatment system, has been applied. The purpose is the thermodynamic and the exergo-economic evaluation of a cogeneration of heat and power cycle (CHP), considering the three objective functions of first and second law efficiencies, and the total specific cost of the system. The combined heat and power plant was fueled by the mixture of natural gas and biogas. A parametric investigation was conducted to assess the influences of decision parameters on the performance and the total cost rate of the cycle. The results showed that in the absence of biogas (high mixing ratios) the irreversibilities of the combustion chamber and the heat recovery steam generator are high, and at low mixing ratios those of the combustion chamber and the anaerobic digester are significant. The irreversibility of the combustion chamber at all the mixing ratios is the highest value. The exergy efficiency of the cycle is increased from 46.94% in the case of pure biogas to 50.64% in the case of pure natural gas. Also, the total specific cost of the system is increased from 66.7 $/MWh in the case of pure natural gas to 98.71 $/MWh in the case of the pure biogas. Dual fuel GT-CHP is an interesting option to access for high efficiency, high flexibility and plant reliability at low cost in comparison to only biogas systems, facilitating a blend of fossil fuel and renewable resource systems.

1. Introduction

The world is facing with the shortage or the absence of fossil fuel resources, which is currently one of the major problems in the world energy sector, especially in the developing countries. Due to the shortage of fossil fuels and their significant contribution on air pollution, climate changes, and the emission of the greenhouse gases (GHGs), renewable fuels are getting more attention to be used instead of fossil fuels. Biofuel is one of these renewable fuels. Gaseous biofuels consist of two different kinds of gas, namely; syngas and biogas. Both syngas and biogas are originated from biomass, the former by gasification and the latter by anaerobic digestion process [1]. The main reason for the GHGs is the breakdown of the balanced carbon cycle to the linear carbon flow into the nature, due to fossil fuel usage. However, the carbon released to atmosphere, due to the anaerobic digestion process of biomasses, is within the environment carbon cycle. Therefore does not add any carbon to the environment. In addition to biogas, anaerobic digestion of biomass produces a byproduct named digestate which can be used as agricultural fertilizer. This process can be considered as a solution for the disposal of the organic fraction of municipal solid waste (OFMSW) in the urban areas. Therefore this technology can be the best tool for improving life, livelihoods, and health in the developing world [2].

Also, since the waste production is being raised continuously, one of the best solutions to solve this problem and get rid of wastes is to convert them to biogas. Therefore, growing attention is being paid to the alteration of waste into beneficial resources. The produced biogas can be used as fuel in combined heat and power plants (CHP). The gas turbines are fuel flexible in terms of the consumed fuel in comparison to other heat engines, therefore, they are suitable candidates for utilizing energy which is released due to the combustion of biogas [3].

In order to investigate the effects of using biogas as a fuel and get rid

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Nomenclature

BL	life time (year)
c_F	average costs associated with fuel ($/GJ)
c_p	average costs associated with product ($/GJ)
C	cost per unit of exergy ($/kJ)
$\dot{C}_D$	cost rate of exergy destruction ($/h)
c_f	cost of fuel per unit of energy ($/MJ)
$\dot{C}_{Loss}$	cost rate of exergy loss ($/h)
$\dot{E}$	exergy rate (kJ/s)
$\dot{E}_D$	exergy destruction rate (kJ/s)
f	exergo-economic factor (%)
h	specific enthalpy (kJ/kg)
LHV	lower heating value (MJ/kg)
HHV	higher heating value (MJ/kg)
$\dot{m}$	mass flow rate (kg/s)
M	molecular mass (kg/kmol)
NPV	net present value ($)
PEC	purchase equipment cost of component ($)
r	relative cost difference (%)
R	revenue
T	temperature (K)
$\dot{Z}$	capital cost rate ($/h)
W	moisture content (kg/kmol)

Greek symbols

η	efficiency
η_{GT}	gas turbine isentropic efficiency
η_{AC}	air compressor isentropic efficiency
η_T	turbine isentropic efficiency
η_{Fc}	fuel compressor isentropic efficiency
ε	exergy efficiency or effectiveness
φ	maintenance factor
Δ	difference

Subscripts and superscripts

0	dead state
0,1,2,3..	cycle locations
	air
AC	air compressor
AD	anaerobic digestion
APH	air preheater
CC	combustion chamber
cen	centrifuge
Cond	condenser
CRF	capital recovery factor
D	destruction
e	exit condition
eco	economizer
ele	electricity
env	environment
Evap	evaporator
FC	fuel compressor
Fd	feedstock
fuel	fuel
GT	gas turbine
GM	gross margin
h	hour
he	heat
HRSG	heat recovery steam generator
HX	heat exchanger
i	inlet condition or interest rate
IHE	internal heat exchanger
k	component
Mc	mixing chamber
mr	mixing ratio
p	price
PB	payback period
PP	pinch point
R_{AC}	compressor pressure ratio
tur	turbine
VS	volatile solid

of wastes, researchers have examined different works. Previous studies have shown that substituting fossil fuels by biogas is a helpful method to reduce greenhouse gases' emissions. Since the disadvantage of methane as a greenhouse gas is much more than that of carbon dioxide, so the use of biogas instead of natural gas mitigates the greenhouse gases' emissions in the atmosphere [4–7]. Therefore, many attempts have been done to replace natural gas with biogas. As an instance, the goal for the contribution of electric power, produced by biogas, is 80 TWh by 2020 in the EU [8]. In this regard, Starr et al. [9] studied the substitution of the upgraded biogas produced from OFMSW with natural gas to reduce the carbon dioxide in Austria, Italy and Spain. The results showed that the amount of biogas, produced from OFMSW, is too low. However, applying the upgraded biogas can reduce natural gas consumption and consequently the greenhouse gases' emission in these countries.

It has been shown that using biogas can result in positive economic effects. Especially when it has been used in a cogeneration system in which the remaining energy in the exhaust gases is exploited [3,4,10,11]. Brizi et al. [10] examined the applicability of biogas and natural gas as fuel in the cogeneration systems from the economic and the energetic points of view. They concluded that in a typical payback period of 5 years, the use of biogas is more economical than natural gas utilization. In the meanwhile, the use of natural gas causes to more energy dissipation compared to biogas. The economic assessment of a cogeneration system using biogas and natural gas were studied in other works and the results showed that the payback period significantly depends on the mixing ratio of biogas and natural gas and the heat sale ratio. As well as, sensitivity analysis revealed that the impact of economic factors on cogeneration systems continuously increases by rising the natural gas percentage in the mixture. Likewise, the cost of electricity and the cost of heat enhances by increasing the percentage of natural gas and by reduction of heat sale ratio [3,10].

In another study done by Kim et al. [11] the performance of a gas turbine cogeneration system fueled by biogas was evaluated. They showed that regardless of operating restrictions on the compressor and turbine, power output increases when methane is replaced by biogas. Also, the amount of heat recovery in steam generator rises when methane content decreases.

The other approved advantage of replacing natural gas by biogas in cogeneration systems is that the performance of the plant fueled by biogas, compared with the systems fueled by natural gas, has been increased in terms of exergy efficiency [6,7].

The cogeneration of power and heat not only is a better way to save energy compared to simple systems, also the economic profit of cogeneration systems is more than that of combined cycles. For example, co-generation systems in terms of net present value is more beneficial compared to the combined cycles. Additionally, there is a strong relation between the income made by selling the heat and the factory

economy [1,5–7]. Camporeale et al. investigated the cogeneration systems of heat and power fueled by the hybrid natural gas and biomass. The results showed that energy conversion efficiency and primary energy saving decreases and investment cost increases by increasing the biomass input rate. Also GT-CHP prepares an attractive option for mixing a renewable resource with a fossil fuel system that is reliable, flexible, efficient, and may be also cost effective in energy demand scenarios [12].

In addition to the OFMSW, there are so many other organic biomasses which can be a good source of energy. Mirmasoumi et al. have studied a combined cooling, heat, and power generating system (CCHP) integrated to a municipal wastewater treatment plant, fueled by biogas. The feedstock fed to their suggested CCHP plant was municipal sewage sludge. They have found that thermal pretreatment of the sewage sludge prior to the digestion process and digestion temperature rise can improve the exergo-economic features of the plant. Also they concluded that compared to mesophilic non-treated digestion of the sewage sludge, thermal pretreatment at 90 °C for 30 min, and thermophilic digestion were able to increase the biomethane productivity of the sewage sludge by 59.82%, and 160.8%, respectively [13]. Wresta et al. carried out an economic analysis on biogas production from cow manure and its usage to generate electricity for a small-scale farm application. Results of the analysis showed that in addition to 2 kW of electricity extraction from cow dung, the residual sludge of the digester can be used as fertilizer that is economically considerable [14]. Also, to enhance the efficiency of biogas production, some solutions have been proposed. For instance Poulsen et al. [15] examined the biogas production potential of different agro products and some types of solid manure. They concluded that the co-digestion of the crops with raw pig manure could result in considerable increase in biogas yield. In another work, it was shown that production potential of biomethane from food waste is between 0.467 and 0.529 $m^3_{methane}/kg_{VS}$. This amount of methane can offset 2.8% of the fuel consumed for transportation in Ireland in 2012 [16]. In another study, Pantaleo et al. thermo-economically appraised a micro gas turbine (100 kWe) fed by the blend of natural gas and biogas. The results indicated that thermal efficiency increases from 38% at pure biomass to 46% at pure natural gas, also electric efficiency increases from 19% to 30% at pure natural gas [17].

Notwithstanding the numerous amount of research work on the biogas fueled cogeneration cycles, there are few studies in the literatures analyzing a biogas fueled cogeneration system by exergo-economic method. The purpose of this work was doing an extensive energy, exergy, and exergo-economic analysis of the suggested AD-CHP plant fueled by the mixture of biogas and natural gas at various mixing ratios. Municipal solid waste was fed to the plant as the feedstock and biogas fuel was considered to be produced from the anaerobic digestion process of OFMSW. The three objective functions of first and second law efficiencies, and the total specific cost of the system were considered to evaluate the influence of co-firing of biogas and natural gas on the economics and efficiencies of the plant. In addition to these objective functions, the payback period for different mixing ratios was appraised, and the major exergy destructing components of the AD-CHP plant were identified. Finally, sensitivity analysis was conducted to identify the important parameters of the cycle.

2. System description

The general layout of the cogeneration system and anaerobic digester has been provided in Fig. 1. As shown in Fig. 1, feedstock (municipal solid waste) at ambient temperature and pressure comes into the digester. It should be mentioned that, the organic fraction of municipal solid waste is collected and crashed and then in a pre-tank is mixed with water to produce a slurry which is considered as the feedstock of the digester. The components of the feedstock was considered to be 90% moisture and 10% solid materials and it was assumed that 0.95 of the solid materials is volatile solid. A well designed digester is able to remove at least 70% of the volatile solid [18].

According to the temperature range in which the microorganisms have been growth, anaerobic digesters are classified to three groups, namely psychrophilic ($T < 20$), mesophilic ($35 < T < 40$), and thermophilic ($55 < T < 60$). In this work, it was assumed that the digestion process is mesophilic and the produced biogas is at 40 °C and atmospheric pressure.

The general reaction of the anaerobic decomposition of OFMSW in

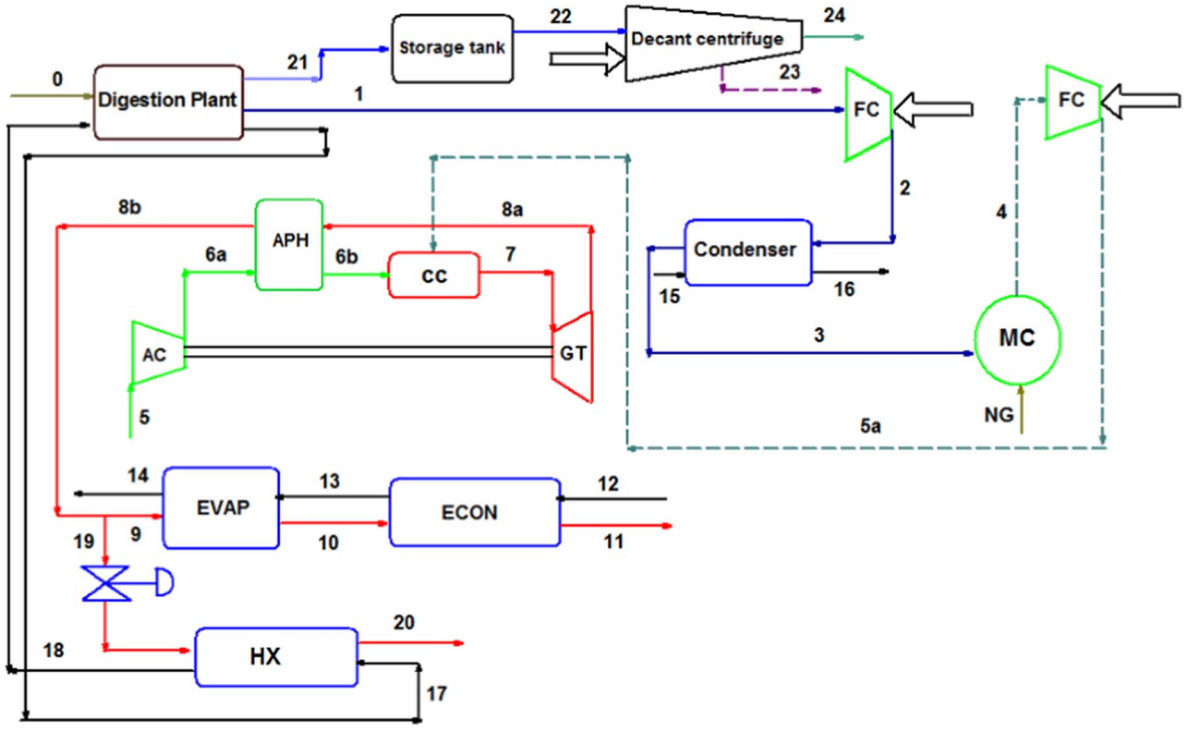

Fig. 1. Schematic diagram of the cogeneration system fueled by biogas and natural.

the digester is as follow [19]:

$$C_cH_HO_oN_NS_S + \frac{1}{4}(4c-h-2o+3n+2s)H_2O \rightarrow \frac{1}{8}(4c-h+2o+3n+2s)CO_2 + \frac{1}{8}(4c+h-2o-3n-2s)CH_4 + nNH_3 + sH_2S \quad (1)$$

The atomic weight and the weight contribution of each element in the dry solids are given in Table 1. Also, it was assumed that the components of the produced biogas are 35% carbon dioxide and 65% methane in mole basis [20].

The amounts of hydrogen sulfides and siloxane are negligible. Biogas and natural gas components and their heating values are listed in Table 2.

According to the mentioned assumptions, the chemical formula of the waste is found to be $C_{100}H_{196}O_{38}$. Based on Fig. 1, biogas at atmospheric pressure (1) enters to the compressor where it is compressed to 5 bar (2), then it is cooled down in the condenser to 30 °C (3). In the case of co-firing, natural gas is added to the biogas in the mixing chamber which is located between the fuel compressors (4). The fuel then passes through a secondary compressor and its pressure reaches to 27 bar (5a).

Air at the temperature of 25 °C and atmospheric pressure (5) enters into the air compressor and its pressure rises up to 10 bar (6a). After leaving the air compressor, the air enters the preheater and leaves it with the temperature of 577 °C (6b). Then, it is mixed and burned with the fuel coming from the secondary compressor in the combustion chamber. Combustion products enter to the turbine at the temperature of 1252 °C (7) and consequently, the turbine produces 30 MW of power. After leaving the turbine, exhaust gases enter the preheater (8a), and then, pass through the heat recovery steam generator (9). On the other side, water at the pressure of 35 bar and the temperature of 25 °C enters the economizer (12) and leaves the evaporator as saturated steam (14). Also a part of the exhaust gasses leaving the combustion chamber (19) can be applied to meet the heat demand of the digester through a heat exchanger (18). After digestion, the digestate enters to the storage tank (21), then it is sent to the decanting centrifuge for dewatering (22) till the liquid (23) and solid (24) are separated so that the solid part has a little moisture content [21,22].

3. Modeling

At this article, it was assumed that the processes were in steady state and the changes of the kinetic and potential energies were neglected.

3.1. Energy analysis

To analyze an energy system, mass and energy balance equations can be written as follows [23]:

$$\left(\frac{dm}{dt}\right)_{cv} = \sum_{in}\left(\frac{dm}{dt}\right) - \sum_{out}\left(\frac{dm}{dt}\right) \quad (2)$$

$$\left(\frac{dE}{dt}\right)_{cv} = \sum_{cv}\left(\frac{dQ}{dt}\right) - \sum_{cv}\left(\frac{dW}{dt}\right) + \sum_{in}\left(\frac{dm}{dt}\right)_{in} h_{in} - \sum_{out}\left(\frac{dm}{dt}\right)_{out} h_{out} \quad (3)$$

The energy balance of the system components are provided in Table 3.

3.2. Exergy analysis

Exergy balance equation for each of the components in the system is written as follow:

$$\sum_{in}\dot{E}_i = \sum_{out}\dot{E}_j + \dot{I} \quad (4)$$

where $\sum_{in}\dot{E}_i$ is the summation of the exergy rate entering to the components and $\sum_{out}\dot{E}_j$ is the summation of the exergy rate outgoing from the system components. $\dot{I}$ is exergy destruction rate in the system components.

Exergy is divided into thermo-mechanical and chemical parts. Thermo-mechanical exergy of a system is the maximum possible work when a system goes from a specified initial state to the limited dead state through a reversible process. Thermo-mechanical exergy, in the absence of magnetic, electrical, nuclear and surface tension effects, consists of three parts, namely, physical exergy, potential exergy, and kinetic exergy as follows:

$$e_{tot} = e_{th} + e_{ch} \quad (5)$$

$$e_{th} = e_{ph} + e_{pt} + e_{kn} \quad (6)$$

The kinetic and potential exergies are ignored in this work. Thus, the thermo-mechanical exergy is reduced to physical exergy and is defined as follow [24]:

$$\dot{E}_{th} = \dot{E}_{ph} = \dot{m}(h-h_0-t_0(s-s_0)) \quad (7)$$

where h, s and t are specific enthalpy, specific entropy and temperature, respectively. Subscript 0 is referred to the environmental conditions. Chemical exergy can be written as below [25]:

$$\bar{e}^{CH} = \sum x_k \bar{e}_k^{CH} + \bar{R}T_0 \sum x_k \ln x_k \quad (8)$$

where $\bar{e}_k^{CH}$ is the chemical exergy, and x_k is the mole fraction of the k_{th} gas in the environmental gas phase and $\bar{R}$ is the universal gas constant.

The input parameters of the cogeneration system are summarized in Table 4.

All the assumptions in the present work are as follows:

- Air contains 77.48% nitrogen, 20.59% oxygen, 0.019% water vapor and 0.0003% carbon dioxide.
- The amount of heat loss in the combustion chamber is assumed to be 2% [20].
- The isentropic efficiencies of the compressors and turbine are 86%.
- The pressure drops in the preheater at air side and gas side are 5% and 3%, respectively.
- Pressure drop in the combustion chamber is considered to be 5%.
- Pressure drop in the heat recovery steam generator is assumed to be 5%.
- Air, the components of the biogas, natural gas and the combustion products are considered as ideal gas [26].
- Heat losses in the turbines and the compressors are considered to be negligible.

3.3. Digester

The chemical exergy of the feedstock is defined as below [27]:

$$e_{ch}^{fd} = \beta \times LHV_{fd} \quad (9)$$

where β is calculated through the following equation for solid fuels [28,29]:

Table 1
Atomic weight and weight contribution of each element in the dry state of feedstock.

Component	Atomic weight	Weight contribution
Carbon	12	0.518
Hydrogen	1	0.08471
Sulfur	32	0.0007
Oxygen	16	0.2625
Nitrogen	14	0.01008
Ash	75	0.1245

Table 2
Natural gas and biogas properties [3].

	Natural gas	Biogas
CH_4 (mole fraction %)	100	65
CO_2 (mole fraction %)	–	35
Lower heating value (kJ/kg)	49,303	20,198

Table 3
Energy balance of system components.

Component	Mass and/or Energy balances
Digestion plant	$\dot{Q}_{loss} + \dot{Q}_{feedstock} + \dot{Q}_{Hot\ water} = 0$ $\dot{W} = 0$ $\dot{m}_0 = \dot{m}_{21} + \dot{m}_1$
Storage tank	$h_{21} = h_{22}$ $\dot{m}_{21} = \dot{m}_{22}$
Centrifuge	$\dot{m}_{22} = \dot{m}_{23} + \dot{m}_{24}$ $\dot{m}_{22}\bar{h}_{22} + \dot{W}_{Cen} = \dot{m}_{23}\bar{h}_{23} + \dot{m}_{24}\bar{h}_{24}$
Fuel compressor (1)	$\dot{W}_{fk} = \dot{m}_{biogas}(\bar{h}_{2a}-\bar{h}_1)/M_{biogas}$ $\eta_{fk} = (\bar{h}_{2s}-\bar{h}_1)/(\bar{h}_{2a}-\bar{h}_1)$
Condenser	$\frac{\dot{m}_{water}(\bar{h}_{15}-\bar{h}_{16})}{M_{water}} = \frac{\dot{m}_{biogas}(\bar{h}_3-\bar{h}_2)}{M_{biogas}}$ $\dot{Q}_{condenser} = \frac{\dot{m}_{water}(\bar{h}_{15}-\bar{h}_{16})}{M_{water}}$
Mixing chamber	$\dot{m}_{biogas} + \dot{m}_{NG} = \dot{m}_{Gas}$ $\dot{m}_{biogas}\bar{h}_3/M_{biogas} + \dot{m}_{NG}\bar{h}/M_{NG} = \dot{m}_{gas}\bar{h}_4/M_{gas}$
Fuel compressor (2)	$\dot{W}_{fc} = \dot{m}_{gas}\frac{(\bar{h}_{5a}-\bar{h}_4)}{M_{biogas}}$ $\eta_{fc} = (\bar{h}_{5s}-\bar{h}_4)/(\bar{h}_{5a}-\bar{h}_4)$
Air compressor	$\dot{W}_{Ac} = \dot{m}_{air}\frac{(\bar{h}_{6a}-\bar{h}_5)}{M_{air}}$ $\eta_{Ac} = (\bar{h}_{6s}-\bar{h}_5)/(\bar{h}_{6a}-\bar{h}_5)$
Air preheater	$\dot{m}_{air}\frac{(\bar{h}_{6a}-\bar{h}_{6a})}{M_{air}} = \dot{m}_{pr}\frac{(\bar{h}_{8a}-\bar{h}_{8b})}{M_{pr}}$ $\dot{Q}_{Airpreheater} = \frac{\dot{m}_{pr}(\bar{h}_{8a}-\bar{h}_{8b})}{M_{pr}}$
Combustion chamber	$\dot{Q}-\dot{W} + \dot{m}_{air}\frac{(\bar{h}_{6a})}{M_{air}} + \dot{m}_{gas}\frac{(\bar{h}_{5a})}{M_{gas}} = \dot{m}_{pr}\frac{(\bar{h}_{pr})}{M_{pr}}$ $\dot{Q} = -0.02\dot{m}_{fuel}LHV_{fuel}$ $\dot{W} = 0$
Gas turbine	$\dot{W}_{GT} = \dot{m}_{Pr}\frac{(\bar{h}_{8a}-\bar{h}_7)}{M_{Pr}}$ $\eta_{GT} = (\bar{h}_{8a}-\bar{h}_7)/(\bar{h}_{8s}-\bar{h}_7)$
Evaporator	$\dot{m}_9\frac{(\bar{h}_9-\bar{h}_{10})}{M_{pr}} = \dot{m}_{steam}\frac{(\bar{h}_{14}-\bar{h}_{13})}{M_{steam}}$ $\dot{Q}_{evaporator} = \frac{\dot{m}_9(\bar{h}_{8b}-\bar{h}_9)}{M_{pr}}$
Economizer	$\frac{\dot{m}_9(\bar{h}_{11}-\bar{h}_{10})}{M_{Pr}} = \frac{\dot{m}_{steam}(\bar{h}_{12}-\bar{h}_{13})}{M_{steam}}$ $\dot{Q}_{economizer} = \frac{\dot{m}_9(\bar{h}_{10}-\bar{h}_{11})}{M_{pr}}$
Heat exchanger	$\frac{\dot{m}_{19}(\bar{h}_{19}-\bar{h}_{20})}{M_{Pr}} = \frac{\dot{m}_{water}(\bar{h}_{18}-\bar{h}_{17})}{M_{water}}$ $\dot{Q}_{heatex} = \frac{\dot{m}_{19}(\bar{h}_{19}-\bar{h}_{20})}{M_{pr}}$ $\dot{m}_9 + \dot{m}_{19} = \dot{m}_{Pr}$

$$\beta = \frac{1.044 + 0.16\frac{H}{C} - 0.34\frac{O}{C}\left(1 + 0.0531\frac{H}{C}\right)}{1 - 0.4142\frac{O}{C}} \quad (10)$$

and LHV_{fd} (kJ/kg) is the lower heating value of the feedstock and is calculated using the following equation [21,30,31]:

$$HHV = (349.1(C) + 1178.3(H) - 103.4(O) + 1.005(S) - 15.1(N) - 21.1ASH) \quad (11)$$

Table 4
Parameters used in the modeling.

Parameters	Value
T_0 (K)	298.15
P_0 (kP)	100
T_1 (K)	313.15
$\dot{W}_{net}$ (MW)	30
mr (%)	0–100
R_P	6–15
η_{AC}	0.72–0.89
η_{GT}	0.78–0.91
T_{6b} (K)	800–880
T_7 (K)	1440–1525
$\Delta T_{pp,HRSG}$ (K)	8–30

$$LHV = HHV - (9\ H + H_2O)h_{fg} \quad (12)$$

where C, H, O, N, S, and Ash are mass percentages of carbon, hydrogen, oxygen, nitrogen, sulfur and ash in the dry solid fuel. Also, h_{fg} is the enthalpy of vaporization of water.

The moisture content per mole of biomass can be expressed in terms of mass based moisture content as follows [9]:

$$W = \frac{M_{biomass} * M_w}{18 * (1 - M_w)} \quad (13)$$

where M_w is the mass fraction of moisture in biomass.

In order to calculate energy conservation that happens in the anaerobic digester, it is required to provide two sources of heat requirement:

- Heat loss from the walls, the floor and the roof of the digester
- Heat to bring feedstock material up to the digestion process temperature

3.3.1. Heat losses

The conductive heat dissipation rate for anaerobic digestion is given in the following equation:

$$\dot{Q}_{loss} = U_{wall}A_{wall}\Delta T_{wall} + U_{roof}A_{roof}\Delta T_{roof} + U_{base}A_{base}\Delta T_{base} \quad (14)$$

$$\Delta T_{wall} = (T_{digestion} - T_{out}) \quad (15)$$

$$\Delta T_{roof} = (T_{digestion} - T_{roof}) \quad (16)$$

$$\Delta T_{base} = (T_{digestion} - T_{base}) \quad (17)$$

where

U = Overall heat transfer coefficient, W/m^2 K
A = Cross-sectional area through which heat loss occurs, m^2
ΔT = Temperature drop across surface, K

The conduction heat transfer coefficients for anaerobic digestion system is given in Table 5, and T_{base} is the soil temperature between the ground and the depth of the digester base, which is considered 285 K.

3.3.2. Heating the feedstock

The feedstock added to the digester must be brought up to the operating temperature of the digester. The required heat for this process is

Table 5
Conduction heat transfer coefficients for digestion system.

Heat transfer coefficients	W/(m^2 K)
K_{wall}	1.53
K_{base}	0.63
K_{roof}	3.31

found by the following equation:

$$\dot{Q}_{\text{feedstock}} = C \times \dot{M} \times \Delta T \tag{18}$$

In which:

C = Feedstock specific heat capacity (kJ/(kg·K))
$\dot{M}$ = Feedstock mass flow rate (kg/s)
ΔT = Temperature difference (K)

3.4. Combustion chamber

It is assumed that complete combustion in accordance with the following reaction occurs inside the combustion chamber [32].

$$\begin{aligned} &n_1CO_2 + n_2CH_4 + n_3(0.2059O_2 + 0.7748N_2 + 0.0003CO_2 + 0.019H_2O) \\ &\rightarrow (n_1 + n_2 + 0.0003n_3)CO_2 + (2n_2 + 0.019n_3)H_2O + 0.7748n_3N_2 \\ &\quad + (0.2059 - 2n_2)O_2 \end{aligned} \tag{19}$$

Here, the coefficients n_1, n_2, are the carbon dioxide and methane mole numbers in the mixture of biogas and natural gas, respectively. Also, n_3 is the mole number of air required for combustion of fuel mixture. The mixing ratio of natural gas and biogas is defined as Eq. (20) [3].

$$mr\ (\%) = \frac{\dot{m}_{\text{NG}} LHV_{\text{NG}}}{(\dot{m}_{\text{NG}} LHV_{\text{NG}} + \dot{m}_{\text{bio}} LHV_{\text{bio}})} * 100 \tag{20}$$

So for pure biogas, the mr becomes 0%, and for pure natural gas it equals to 100%.

3.5. Economic and exergo-economic analysis of the system

Combination of the second law of thermodynamics and economic concepts is known as Exergo-economics. It is a powerful tool for optimization of energy systems [33]. In this regard, various techniques including, the exergy cost theory [34], the average cost approach [35] and the specific exergy costing (SPECO) [33,36,37] are well-known. The SPECO method has advantages of less calculation time and an easy and straightforward scheme. Hence, it is used for exergo-economic analysis of the system in this work. The SPECO method has three steps: firstly, all energetic and exergetic flows are determined in the boundaries of the considered components, then each flow is assigned as fuel or product and finally, the cost balance and auxiliary equations are written based on the following equations for each component [23,33,36,38,39]:

$$\sum_{\text{j}} \dot{C}_{\text{j,k}} + \dot{C}_{\text{w,k}} = \dot{C}_{\text{q,k}} + \sum_{\text{i}} \dot{C}_{\text{i,k}} + \dot{Z}_{\text{k}} \tag{21}$$

Here $\dot{C}$ is the cost rate, $\dot{Z}$ is the cost rate associated with the capital investment cost and cost rate related to operation and maintenance of the k_{th} component. The subscripts of i and j represent entry and exit flows of the system, respectively. $\dot{C}_{\text{w,k}}$, $\dot{C}_{\text{q,k}}$ show the cost rate associated with the work and heat transfer.

In the cost balance equation, the cost rates of exergy streams are defined as [23,40]:

$$\dot{C}_{\text{i}} = c_{\text{i}} \dot{E}_{\text{i}} \tag{22}$$

$$\dot{C}_{\text{j}} = c_{\text{j}} \dot{E}_{\text{j}} \tag{23}$$

$$\dot{C}_{\text{q}} = c_{\text{q}} \dot{E}_{\text{q}} \tag{24}$$

$$\dot{C}_{\text{w}} = c_{\text{w}} \dot{W} \tag{25}$$

where c_i, c_j, c_q and c_w are cost per the unit of exergy in \$/GJ. $\dot{Z}_k$, is the sum of the costs rate associated with capital investment cost ($\dot{Z}_k^{CI}$), and operating and maintenance cost ($\dot{Z}_k^{OM}$) [23,41]:

$$\dot{Z}_{\text{k}} = \dot{Z}_{\text{k}}^{\text{CL}} + \dot{Z}_{\text{k}}^{\text{OM}} = CRF * \frac{\phi_{\text{r}}}{\text{N}} * PEC_{\text{k}} \tag{26}$$

Here, PEC_k, is the purchased equipment cost of the k^{th} component (\$), N (h) is the annual operating hours of the system and φ_r is the maintenance factor, which is considered to be 1.06. The capital recovery factor, *CRF*, depends on the interest rate and the estimated equipment life time, and it is defined as below [42–44]:

$$CRF = \frac{i*(1+i)^{bl}}{(1+i)^{bl} - 1} \tag{27}$$

Here i is the interest rates and bl is the useful life of the system. In this work, bl and i are assumed to be 20 year and 0.12, respectively [3,45].

Based on Ref. [26], the unit cost of natural gas, c_f, is 0.009 \$/MJ and the capital investment cost and operating & maintenance cost for digester is calculated as follows [46]:

$$Z_{\text{digester}}^{\text{CI}} = 1.6049 \times x^{(0.5099)}\ (\text{Millions"\$"}) \tag{28}$$

$$Z_{\text{digester}}^{OM} = 264.69 \times x^{(-0.176)}\ (\text{"\$"/ton}) \tag{29}$$

In which, "x" is total solid waste in thousand tons per year.

Also, for all the mixing ratios, the costs of slurry treating with a decanting centrifuge is considered on average 0.88(\$/ton) [22]. Hence, the cost rate of the natural gas conducted into the combustion chamber is [47]:

$$\dot{C}_{\text{f}} = c_{\text{f}} \times \dot{m}_{\text{f}} \times LHV \tag{30}$$

The correlations used for the purchased equipment cost (PEC) for the system components are given in Table 6.

Of course the following formula is used to convert the cost of purchasing equipment from the base year to the reference year.

Table 6
The correlations used for the purchased equipment cost (*PEC*) of the components.

Component	Purchased equipment cost (PEC)	Year	Exchange ratio
Air compressor	$pec_{AC} = \left(\frac{71.10\dot{m}_{air}}{0.9-\eta_{is,c}}\right)\left(\frac{P_{out}}{P_{in}}\right)Ln\left(\frac{P_{out}}{P_{in}}\right)$	1995	$\frac{1267}{1027}$
Air preheater	$pec_{APH} = 4122 \times \left(\frac{\dot{m}_g(h_{8a}-h_{8a})}{u \times \Delta T_{lm,APH}}\right)^{0.6}$	1995	$\frac{1267}{1027}$
Combustion chamber	$pec_{cc} = \frac{46.08 \times \dot{m}_{air}}{(0.995 - \frac{P_4}{P_3})}[1 + \exp(0.018T_4 - 26.4)]$	1995	$\frac{1267}{1027}$
Gas turbine	$pec_{GT} = \frac{479.34 \times \dot{m}_g}{(0.92-\eta_{GT})}\ln\left(\frac{p_4}{p_5}\right)[1 + \exp(0.036T_7 - 54.4)]$	1995	$\frac{1267}{1027}$
Heat recovery steam generator	$pec_{HRSG} = 6570\left[\left(\frac{\dot{Q}_{ec}}{\Delta T_{lm,ec}}\right)^{0.8} + \left(\frac{\dot{Q}_{ev}}{\Delta T_{lm,ev}}\right)^{0.8}\right] + 21276\dot{m}_w + 1184.4\dot{m}_g^{1.2}$	1995	$\frac{1267}{1027}$
Heat exchanger	$pec_{HE} = 190 + 310 \times A_{HE}$	2010	$\frac{1267}{1194}$
Fuel compressor	$pec_{FC} = 91562 \times \left(\frac{W_{FC}}{445}\right)^{0.67}$	1996	$\frac{1267}{1039}$

Table 7
Exergetic cost rate balance and required auxiliary equation(s) for each component of the system (State numbers refer to Fig. 1).

Components	Cost Balance	Auxiliary equations
Digestion plant	$\dot{C}_0 + \dot{C}_{18} + \dot{Z}_0 = \dot{C}_1 + \dot{C}_{17} + \dot{C}_{21}$	$\frac{\dot{C}_{17}}{\dot{E}_{17}} = \frac{\dot{C}_{18}}{\dot{E}_{18}}$ $\frac{\dot{C}_{21}}{\dot{E}_{21}} = \frac{\dot{C}_1}{\dot{E}_1}$
Storage tank	$\dot{C}_{21} + \dot{Z}_{storage} = \dot{C}_{22}$	None
Centrifuge	$\dot{C}_{22} + \dot{C}_{W,Cen} + \dot{Z}_0 = \dot{C}_{24} + \dot{C}_{23}$	$\frac{\dot{C}_{W,Cen}}{\dot{W}_{Cen}} = \frac{\dot{C}_{W,Gt}}{\dot{W}_{GT}}$
Fuel compressor (1)	$\dot{C}_1 + \dot{C}_{W,FC1} + \dot{Z}_{FC1} = \dot{C}_2$	$\frac{\dot{C}_{FC,1}}{\dot{W}_{FC,1}} = \frac{\dot{C}_{W,Gt}}{\dot{W}_{GT}}$
Mixing chamber	$\dot{C}_3 + \dot{C}_{f,NG} = \dot{C}_4$	None
Fuel compressor (2)	$\dot{C}_4 + \dot{C}_{W,FC2} + \dot{Z}_{FC2} = \dot{C}_{5a}$	$\frac{\dot{C}_{FC,2}}{\dot{W}_{FC,2}} = \frac{\dot{C}_{W,Gt}}{\dot{W}_{GT}}$
Air compressor	$\dot{C}_5 + \dot{C}_{W,AC} + \dot{Z}_{AC} = \dot{C}_{6a}$	$\frac{\dot{C}_{W,Ac}}{\dot{W}_{AC}} = \frac{\dot{C}_{W,Gt}}{\dot{W}_{GT}}$
Air preheater	$\dot{C}_{5a} + \dot{C}_{8a} + \dot{Z}_{A,P} = \dot{C}_{8b} + \dot{C}_{5b}$	$\frac{\dot{C}_{8a}}{\dot{E}_{8a}} = \frac{\dot{C}_{8b}}{\dot{E}_{8b}}$
Combustion chamber	$\dot{C}_{5a} + \dot{C}_{6b} + \dot{Z}_{comb} = \dot{C}_7$	None
Gas turbine	$\dot{C}_7 + \dot{C}_{GT} = \dot{C}_{8a} + \dot{C}_{W,GT} + \dot{C}_{W,AC} + \dot{C}_{W,Cen} + \dot{C}_{FC,1} + \dot{C}_{FC,2}$	$\frac{\dot{C}_{8a}}{\dot{E}_{8a}} = \frac{\dot{C}_{8b}}{\dot{E}_{8b}}$
Heat recovery steam generator	$\dot{C}_9 + \dot{C}_{12} + \dot{Z}_{HRSG} = \dot{C}_{11} + \dot{C}_{14}$	$\frac{\dot{C}_9}{\dot{E}_9} = \frac{\dot{C}_{11}}{\dot{E}_{11}}$ $\dot{C}_{12} = 0$
Heat ex	$\dot{C}_{19} + \dot{C}_{17} + \dot{Z}_{HE} = \dot{C}_{18} + \dot{C}_{20}$ $\dot{C}_{19} + \dot{C}_{17} + \dot{Z}_{HE} = \dot{C}_{18} + \dot{C}_{20}$	$\frac{\dot{C}_{19}}{\dot{E}_{19}} = \frac{\dot{C}_{20}}{\dot{E}_{20}}$

Table 8
The definition of the exergy rate of fuel and product, and exergy balance for the components.

Components	Exergy rate of fuel	Exergy rate of product	Exergy balances
Digestion plant	$\dot{E}_0 + \dot{E}_{18} - \dot{E}_{17}$	$\dot{E}_1 + \dot{E}_{21}$	$\dot{E}_0 + \dot{E}_{18} = \dot{E}_1 + \dot{E}_{21} + \dot{E}_{17} + \dot{I}_{DG}$
Storage tank	$\dot{E}_{21}$	$\dot{E}_{22}$	$\dot{E}_{21} = \dot{E}_{22} + \dot{I}_{storage}$
Centrifuge	$\dot{E}_{22} + \dot{W}_{cen}$	$\dot{E}_{24}$	$\dot{E}_{22} + \dot{W}_{cen} = \dot{E}_{23} + \dot{E}_{24} + \dot{I}_{cen}$ $\dot{W}_{cen} = 3.0$ kWh/ton
Fuel compressor (1)	$\dot{W}_{FC,1}$	$\dot{E}_2 - \dot{E}_1$	$\dot{E}_1 + \dot{W}_{FC,1} = \dot{E}_2 + \dot{I}_{FC,1}$
Condenser	$\dot{E}_2 - \dot{E}_3$	$\dot{E}_{16} - \dot{E}_{15}$	$\dot{E}_2 + \dot{E}_{15} = \dot{E}_{16} + \dot{E}_3 + \dot{I}_{cond}$
Mixer chamber	$\dot{E}_3 + \dot{E}_{f,NG}$	$\dot{E}_4$	$\dot{E}_3 + \dot{E} = \dot{E}_4 + \dot{I}_{mix}$
Fuel compressor (2)	$\dot{W}_{FC,2}$	$\dot{E}_{5a} - \dot{E}_4$	$\dot{E}_4 + \dot{W}_{FC,2} = \dot{E}_{5a} + \dot{I}_{FC,2}$
Air compressor	$\dot{E}_{w,Ac}$	$\dot{E}_{6a} - \dot{E}_5$	$\dot{E}_5 + \dot{W}_{AC} = \dot{E}_{6a} + \dot{I}_{Ac}$
Air preheater	$\dot{E}_{8a} - \dot{E}_{8b}$	$\dot{E}_{6b} - \dot{E}_{6a}$	$\dot{E}_{6a} + \dot{E}_{8a} = \dot{E}_{6b} + \dot{E}_{8b} + \dot{I}_{APH}$
Combustion chamber	$\dot{E}_{5a}$	$\dot{E}_7 - \dot{E}_{6b}$	$\dot{E}_{6b} + \dot{E}_{5a} = \dot{E}_7 + \dot{I}_{combustor}$
Gas turbine	$\dot{E}_7 - \dot{E}_{8b}$	$\dot{E}_{GT}$	$\dot{E}_7 = \dot{E}_{8a} + \dot{E}_{GT} + \dot{I}_{GT}$
Evaporator	$\dot{E}_9 - \dot{E}_{10}$	$\dot{E}_{14} - \dot{E}_{13}$	$\dot{E}_9 + \dot{E}_{13} = \dot{E}_{10} + \dot{E}_{14} + \dot{I}_{evap}$
Economizer	$\dot{E}_{10} - \dot{E}_{11}$	$\dot{E}_{13} - \dot{E}_{12}$	$\dot{E}_{10} + \dot{E}_{12} = \dot{E}_{11} + \dot{E}_{13} + \dot{I}_{economizer}$
Heat ex	$\dot{E}_{19} - \dot{E}_{20}$	$\dot{E}_{18} - \dot{E}_{17}$	$\dot{E}_{19} + \dot{E}_{17} = \dot{E}_{20} + \dot{E}_{18} + \dot{I}_{ex}$

$$PEC_{new} = PEC_{ref}\left(\frac{CI_{new}}{CI_{ref}}\right) \tag{31}$$

Here CI, is the cost index. The values of Marshall and Swift cost index are used.

3.5.1. Auxiliary equations

In the components of thermal systems, in the case of k exergy streams exiting the considered component, $k-1$ auxiliary equations are needed for solving the exergo-economic balance equations. The F and P rules can be used for calculation of the exergy destruction cost rate [26,36,48].

By employing the auxiliary equations, a system of linear equations $([\dot{E}_k] \times [c_k] = [\dot{Z}_k])$ are formed based on the developed equation for each component of the cycle. Therefore, by solving these sets of equations, the cost rate of each line can be calculated [49]. Here, $[\dot{E}_k]$ is the matrices of exergy rate which is obtained by the exergy analysis, $[c_k]$ is exergetic cost vector (to be evaluated), and $[\dot{Z}_k]$ is the vector of $\dot{Z}$ values which is obtained by Eq. (26).

Exergy cost rate balance and cost rate relations for the components of the system and also exergy rate of fuel, exergy rate of product and exergy balances for the components of cogeneration system are provided in Tables 7 and 8 respectively. To solve the equations, related to the thermo-economic modeling of the system, Engineering Equation Solver (EES) software has been used.

4. Performance criteria and objective functions

To evaluate the performance of a co-generation system practicability, some criteria are defined as below. The first and the second laws' efficiencies for the system are provided as follows [1,11]:

$$\eta_{th} = \frac{\dot{W}_{GT} - \dot{W}_{FC,1} - \dot{W}_{FC,2} - \dot{W}_{Cen} + \dot{E}_{14} - \dot{E}_{12}}{(\dot{m}_{NG} LHV_{NG} + m_{fd} LHV_{fd})} \tag{32}$$

Table 9
Sale prices of the economic factors.

Economic factors	Prices
Electricity price ($/kWh)	0.1436
Heat price ($/kWh)	0.0738
Fertilizer price ($/ton)	800

Table 10
Comparing the results of this work with that of Ref. [48].

η_I (%)		ϵ (%)		$\dot{m}_{NG}$ (kg/s)		$\dot{m}_{air}$ (kg/s)	
[48]	Present work	[48]	Present work	[48]	Present work	[48]	Present work
53.61	52.57	50.7	50.64	1.7	1.69	94.75	94.76

Table 11
The effect of mixing ratio on the performance of the cycle.

	mr = 0% (Pure biogas)	mr = 25%	mr = 50%	mr = 75%	mr = 100% (Pure NG)
Power generation efficiency (%)	28.86	30.23	31.7	33.29	35.02
First law efficiency (%)	41.82	44.2	46.77	49.56	52.57
Second law efficiency (%)	46.94	47.69	48.55	49.52	50.64
Fuel rate of biogas (kg/s)	4.21	3.15	2.1	1.04	0
Fuel rate of NG (kg/s)	0	0.426	0.851	1.275	1.69
HRSG heat recovery (MW)	35.58	36.95	38.33	39.70	41.07
Hot steam generated (kg/s)	13.21	13.72	14.22	14.73	15.24
Heat supply for digester (MW)	4.5	3.37	2.25	1.123	0
Fuel pretreatment power (MW)	1.23	0.984	0.741	0.5	0.256
Total cost rate of system ($/h)	4385	4235	4028	3725	3066
Total specific cost of system ($/MWh)	98.71	94.5	89.12	81.72	66.7

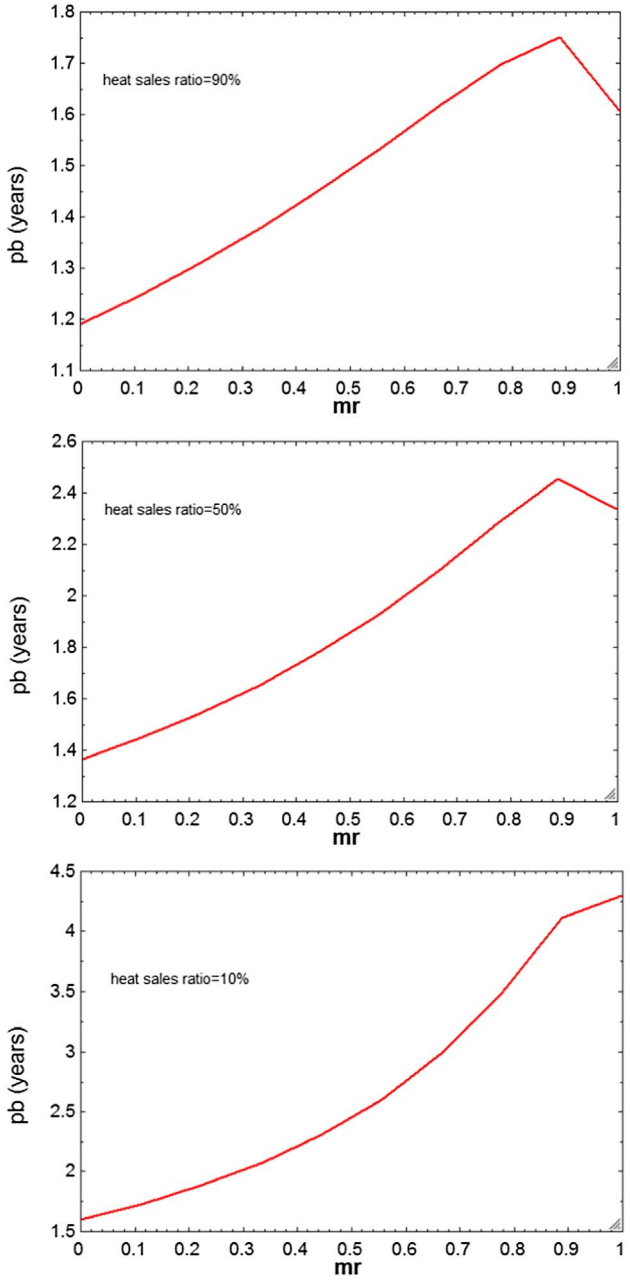

Fig. 2. The effect of mixing ratio on payback period at different heat sales ratios.

$$\varepsilon = \frac{\dot{W}_{GT} - \dot{W}_{FC,1} - \dot{W}_{FC,2} - \dot{W}_{Cen} + \dot{E}_{14} - \dot{E}_{12} + \dot{E}_{24}}{\dot{E}_5 + \dot{E}_{NG} + \dot{E}_{fd}} \tag{33}$$

The electrical efficiency is also defined as below [50]:

$$\eta_{el} = \frac{\dot{W}_{GT} - \dot{W}_{FC,1} - \dot{W}_{FC,2} - \dot{W}_{Cen}}{(\dot{m}_{NG} LHV_{NG} + m_{fd} LHV_{fd})} \tag{34}$$

$$\dot{W}_{GT} = \dot{W}_T - \dot{W}_C \tag{35}$$

And the total system cost rate, $\dot{C}_{system}$, is defined by Eq. (33) [26]:

$$\dot{C}_{system} = \dot{C}_{NG} + \dot{C}_{Fd} + \sum_i \dot{Z}_i \tag{36}$$

Also, the average costs per the exergy unit of fuel and product, and the cost rate of exergy destruction are defined as relations (37), (38) and (39), respectively.

$$c_{f,k} = \dot{C}_{f,k} / \dot{E}_{f,k} \tag{37}$$

$$c_{p,k} = \dot{C}_{p,k} / \dot{E}_{p,k} \tag{38}$$

$$\dot{C}_{D,k} = c_{f,k} \dot{E}_{D,k} \tag{39}$$

Relative cost difference, r_k, and exergo-economic factor, f_k, are defined as below:

$$r_k = (C_{p,k} - C_{f,k}) / C_{f,k} \tag{40}$$

$$f_k = (\dot{Z}_k) / (\dot{Z}_k + \dot{C}_{D,k} + \dot{C}_{l,k}) \tag{41}$$

Payback period (PB) is a parameter presenting the time needed for an investment return. It is obvious that longer PB is not acceptable for an investment regaining.

The payback period for each case study project can be calculated by dividing TDI (total depreciable investment) by the ANCF (annual net cash inflow) [48]:

$$PB = \frac{TDI}{ANCF} \tag{42}$$

$$ANCF = R_{el} + R_h + R_{fertilzer} - (\dot{C}_{fd} + \dot{C}_{NG} + \dot{Z}^{OM}_{overall}) \tag{43}$$

TDI includes the purchased-equipment cost, installation, Piping, Engineering costs, etc. [48]. Also R_{el}, R_h and $R_{fertilzer}$ denote respectively the expected yearly sales (revenues) of electricity, heat and fertilzer which are expressed as follows [3].

$$R_{el} = P_{el}(\dot{W}_{GT} - \dot{W}_{FC,1} - \dot{W}_{FC,2} - \dot{W}_{cen}) * N \tag{44}$$

$$R_h = P_{he} \times \dot{Q}_{HRSG} \times N \tag{45}$$

$$R_{fertilzer} = P_{fertilzer} \times \dot{m}_{24} \times 3.6 \times N \tag{46}$$

Table 12
Exergo-economic analysis for various components of the cycle for pure biogas condition (mr = 0%).

Components	c_f ($/GJ)	c_p ($/GJ)	$\dot{C}_D$ ($/h)	$\dot{C}_{loss}$ ($/h)	$\dot{Z}$ ($/h)	$\dot{Z} + \dot{C}_D$ ($/h)	r (%)	f (%)
Digester	10.03	11.26	415.7	–	86.86	502.56	12.17	17.28
Fuel compressor (1)	19.29	22.74	4.318	–	1.855	6.173	17.35	30.05
Condenser	11.32	56.47	3.176	–	0.298	3.474	398.9	8.58
Fuel compressor (2)	19.29	22.63	5.075	–	2.092	7.176	17.93	29.19
Air compressor	19.29	21.44	145.7	–	64.92	210.62	11.16	30.82
Air preheater	18.3	22.21	185	–	15.21	200.21	21.19	7.59
Combustion chamber	14.68	18.3	1299	66.94	6.266	1305	24.69	0.45
Gas turbine	18.3	19.29	193.2	–	72.62	265.8	5.39	27.32
HRSG	18.3	25.97	333.4	248.1	22.79	356.19	41.87	3.77
Heat ex	18.3	59.79	114.4	109.2	0.1157	114.57	226.7	0.051

Table 13
Exergy-economic analysis for various components of the cycle (mr = 50%).

Components	c_f ($/GJ)	c_p ($/GJ)	$\dot{C}_D$ ($/h)	$\dot{C}_{loss}$ ($/h)	$\dot{Z}$ ($/h)	$\dot{Z} + \dot{C}_D$ ($/h)	r (%)	f (%)
Digester	11.54	12.95	238.5	–	51.55	290	12.17	17.77
Fuel compressor (1)	18.93	22.61	2.11	–	1.164	3.274	19.44	35.5
Condenser	13	64.33	1.819	–	0.150	1.96	394.9	7.65
Fuel compressor (2)	18.93	22.45	3.56	–	1.638	5.198	18.56	31.47
Air compressor	18.93	21.06	146.2	–	66.34	212.5	11.22	31.22
Air preheater	17.88	21.65	182	–	15.61	197.6	21.11	7.89
Combustion chamber	14.37	17.88	1260	65.49	6.402	1266.4	24.39	0.48
Gas turbine	17.88	18.93	191.2	–	73.17	264.3	5.912	27.68
HRSG	17.88	25.36	350.2	249.7	24.52	374.7	41.85	3.92
Heat ex	17.88	58.22	55.45	52.51	0.059	55.50	225.7	0.054

Table 14
Exergy-economic analysis for various components of the cycle (mr = 100%).

Components	c_f ($/GJ)	c_p ($/GJ)	$\dot{C}_D$ ($/h)	$\dot{C}_{loss}$ ($/h)	$\dot{Z}$ ($/h)	$\dot{Z} + \dot{C}_D$ ($/h)	r (%)	f (%)
Fuel compressor (2)	15.88	19.38	1.75	–	1.102	2.85	22.09	38.62
Air compressor	15.88	17.76	125.2	–	67.74	192.9	11.9	35.12
Air preheater	14.87	18.01	152.4	–	16.01	168.4	21.18	9.50
Combustion chamber	11.97	14.87	1043	54.52	6.538	1049	24.18	0.59
Gas turbine	14.87	15.88	161.1	–	73.71	234.8	6.79	31.39
HRSG	14.87	21.16	311.4	213	26.27	337.6	42.34	4.77

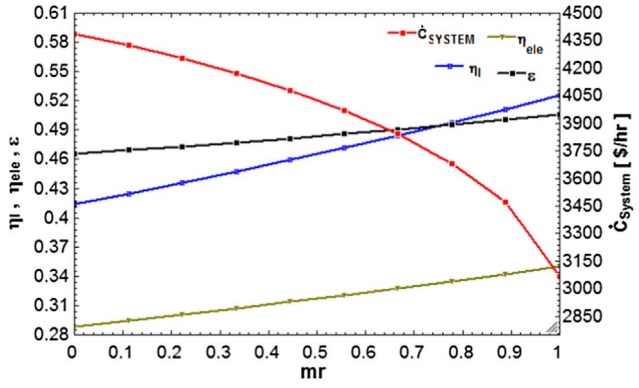

Fig. 3. The effect of mixing ratio on the first and second law efficiencies, electrical efficiency and the total system cost rate.

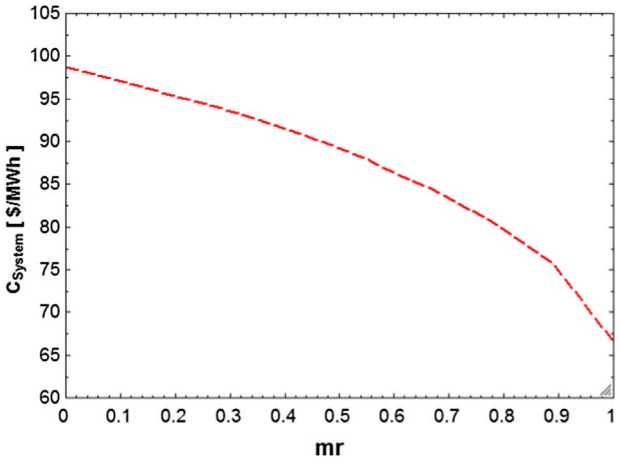

Fig. 4. The effect of mixing ratio on the total specific cost of the system.

where P_{el}, P_{he} *and* $P_{fertilzer}$ represent selling (market) prices of electricity, heat and fertilzer, respectively. The lists of price data are given in Table 9 [51].

5. Results and discussion

In this study, initially the gas turbine cycle fueled by natural gas was validated with Ref. [48]. The first and the second law efficiencies, fuel and air mass flow rates for the cycle are shown in Table 10. It can be seen that there is a good agreement between the results of the current research and those of Ref. [48]. Then, anaerobic digestion process and other components were added to the cycle for production of biogas as the cycle fuel. Finally, the cycle was analyzed in terms of energy, exergy and exergo-economic concepts.

Performance analysis of the cogeneration system for different mixing ratios of biogas and natural gas is shown in Table 11. Gas

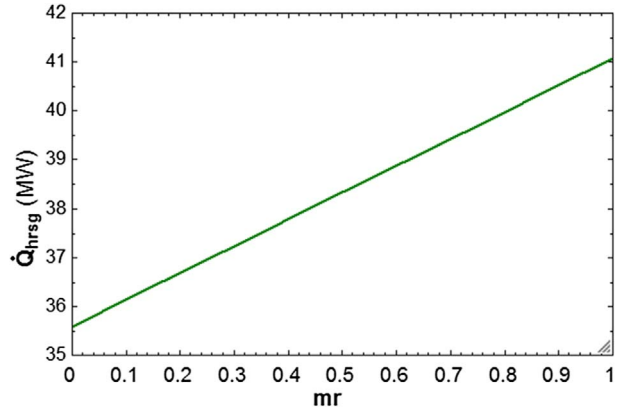

Fig. 5. The effect of mixing ratio on the rate of the generated heat in the HRSG.

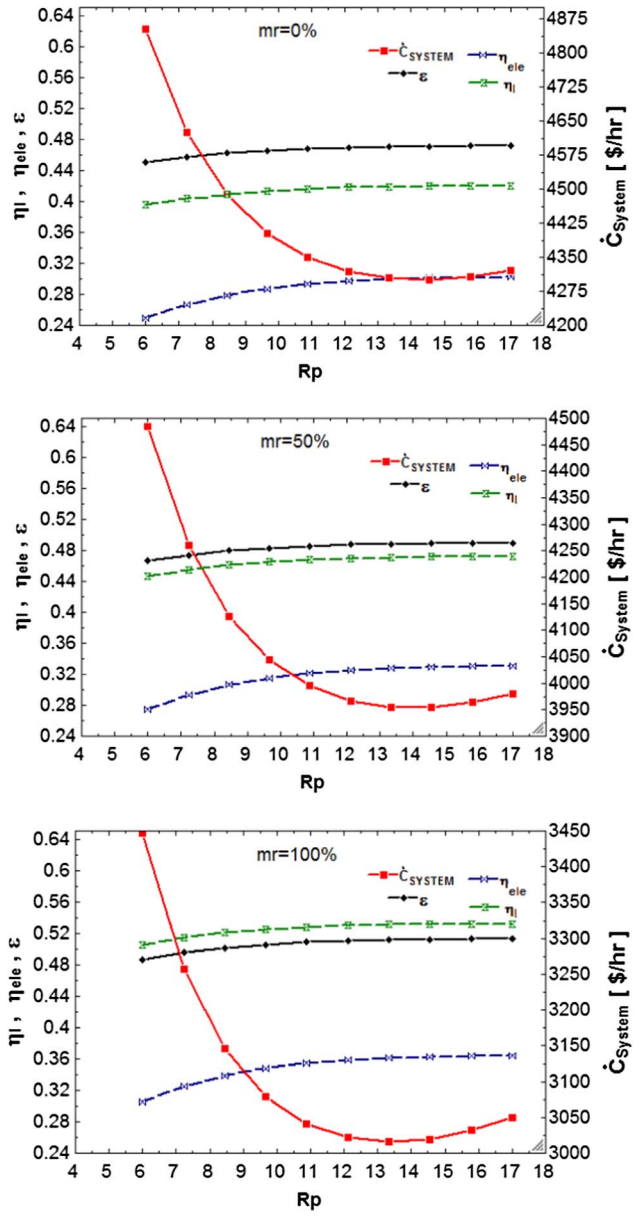

Fig. 6. The effect of air compressor pressure ratio on the first and the second law, and electrical efficiencies, and the total system cost rate at different mixing ratios.

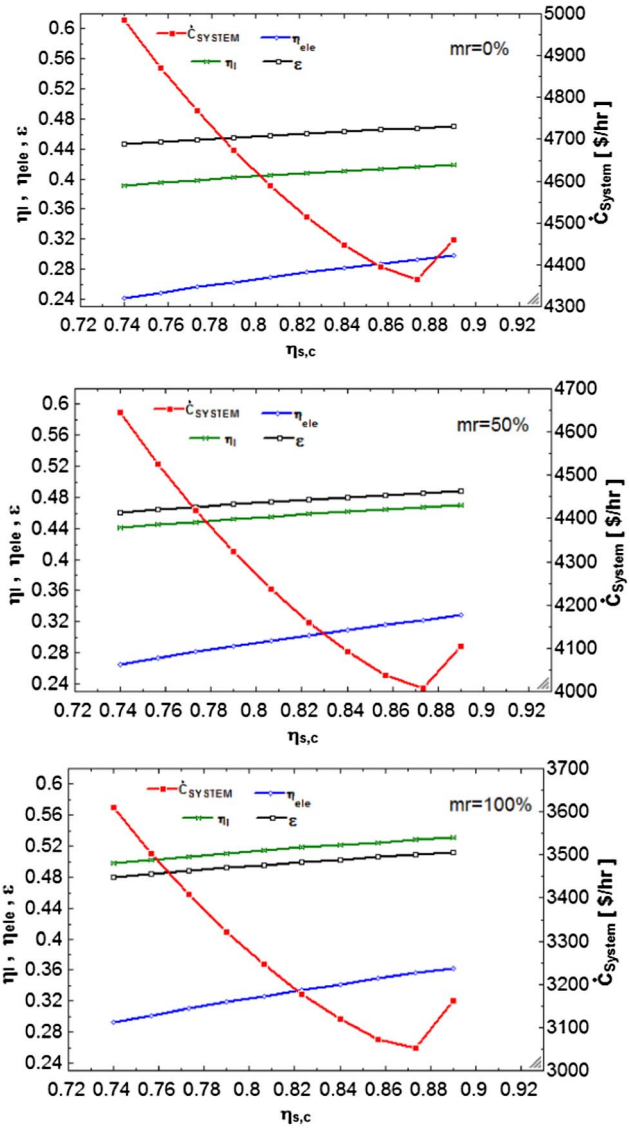

Fig. 7. The effect of air compressor isentropic efficiency on the first and the second law, and electrical efficiencies and the total system cost rate at different mixing ratios.

turbine capacity was kept constant and equal to 30 MW. As it can be seen, the first and the second law efficiencies are decreased from 52.57% to 41.82% and from 50.64% to 46.94%, respectively, and the electrical efficiency is decreased from 35.02% to 28.86% by increasing the percentage of biogas in the mixture.

By increasing the percentage of NG in the fuel mixture, the amount of heat recovered in the heat recovery steam generator is increased from 35.58 MW to 41.07 MW, also this change leads to an increase in the generated steam mass flow rate from 13.21 kg/s to15.24 kg/s.

The effect of the mixing ratio on the payback period is shown in Fig. 2 at various heat sales ratios.

By increasing the mixing ratio, the compost product decreases and the generated heat increases. The interaction of these two factors can be seen in the mixing ratio of around 90%. Certainly, with the increase in the percentage of the heat sales, the payback period will decrease.

5.1. Exergo-economic results

Exergo-economic analysis is one of the best means for evaluating energy systems. Tables 12–14 show the results of exergo-economic analysis for the studied cogeneration system. The results reveal that the combustion chamber, the digester and the gas turbine have the highest

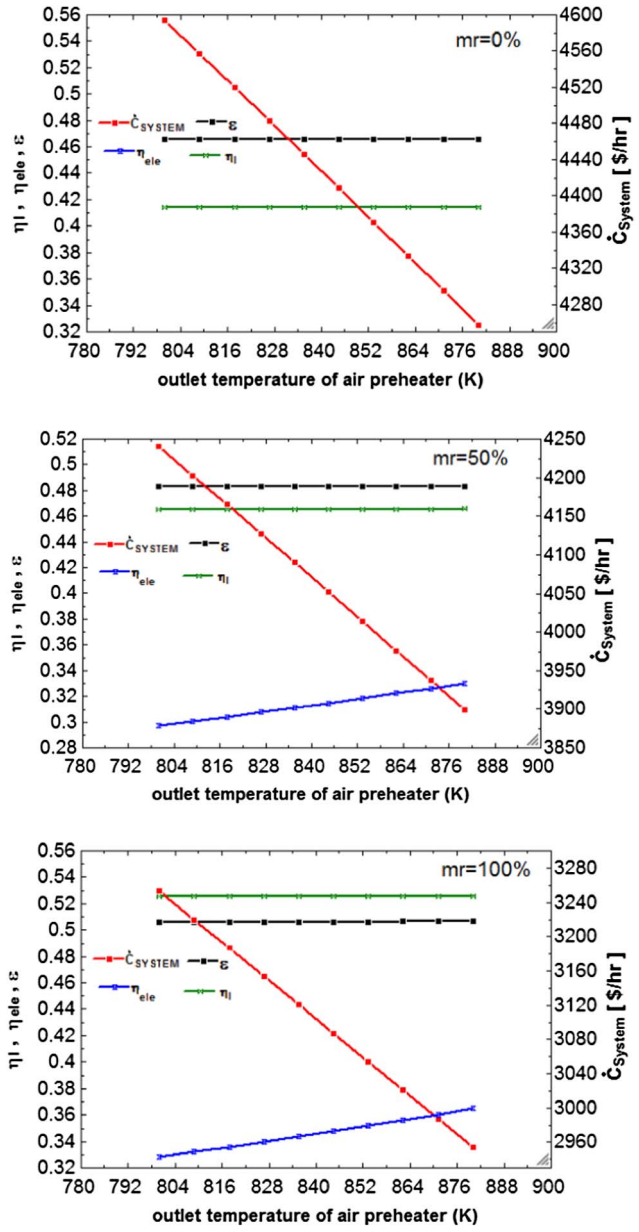

Fig. 8. The effect of air preheater outlet temperature on the first and the second law, and electrical efficiencies and the total system cost rate at different mixing ratios.

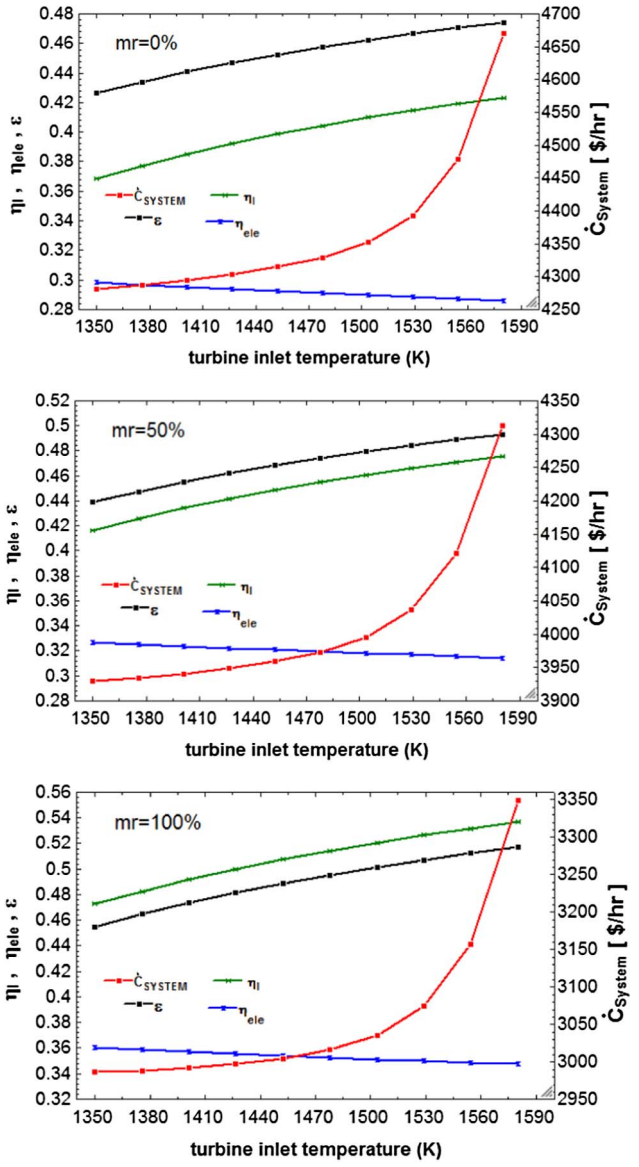

Fig. 9. The effect of gas turbine inlet temperature on the first and the second law, and electrical efficiencies and the total system cost rate at different mixing ratios.

amount of $\dot{Z} + \dot{C}_D + \dot{C}_{Loss}$. Therefore, these are the most important components from the exergo-economic viewpoint. The relative cost difference in the condenser and heat exchanger is high which is due to higher exergy destruction cost rate in these components.

Exergo-economic factor represents the relation between the investment cost rate and the exergy destruction cost rate. Lower amounts of the exergo-economic factor in the combustion chamber and the heat exchanger show that these components have a higher exergy destruction cost rate. Therefore, exergy destruction cost rate must be reduced to improve the efficiency of these components. The higher value of exergo-economic factor in a component shows that the capital cost rate in that component is high and it should be reduced. As it can be inferred from Tables 12–14, the exergo-economic factor of the fuel compressors, the air compressor, and the gas turbine are high.

By increasing the mixing ratio, the total specific cost of the system decreases remarkably. As shown in Table 11, the total specific cost of the system has been decreased from 98.71 $/MWh for the case of pure biogas to 66.7 $/MWh for the case of pure natural gas. Furthermore by increasing the contribution of natural gas, the exergy destruction cost rate of the most of components reduced. In different mixing ratios, according to the exergy destruction cost rate, the combustion chamber, the digester, the heat recovery steam generator, the gas turbine and the air preheater are the most important components of the system. Insofar as, the exergy destruction cost rates of the combustion chamber and the digester are the highest values, respectively.

5.2. Parametric study results

A parametric study was carried out to determine the effects of important design parameters on the system performance. The considered parameters were the mixing ratio of biogas and natural gas (mr), the air compressor pressure ratio (Rp), isentropic efficiency of the air compressor ($\eta_{is,AC}$), the outlet temperature of the air pre-heater, the gas turbine inlet temperature, isentropic efficiency of the gas turbine ($\eta_{is,GT}$), and pinch point temperature difference of the evaporator ($\Delta T_{pp,evaporator}$) to evaluate the impact of each parameter, the rest of the parameters were kept constant.

Figs. 3 and 4 show the effect of mixing ratio variation on the second law efficiency, total cost rate, and the total specific cost of the system.

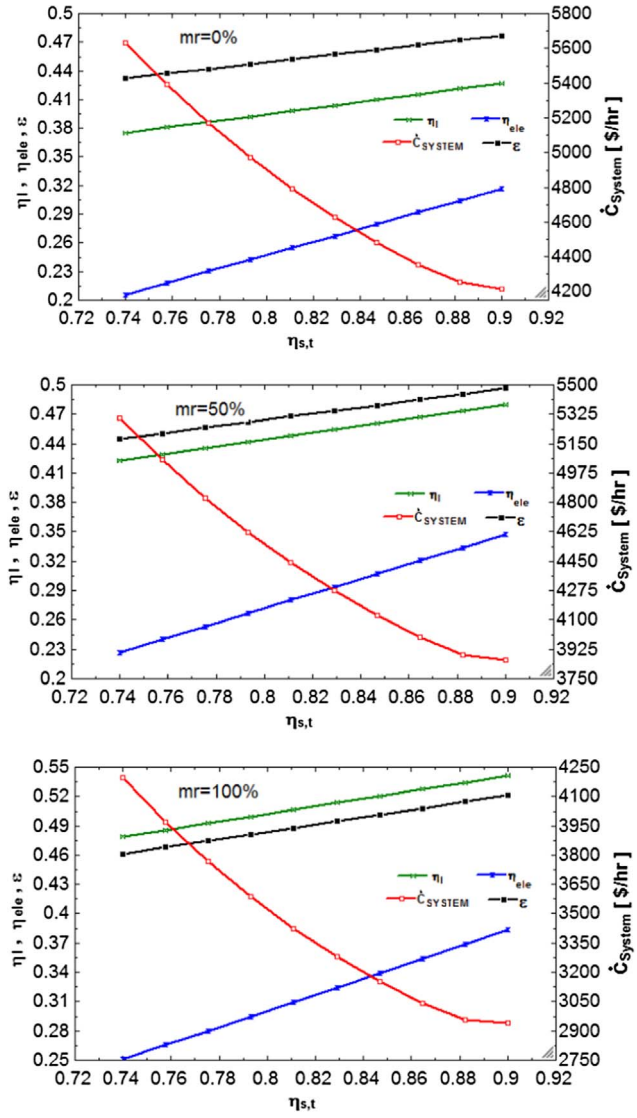

Fig. 10. The effect of gas turbine isentropic efficiency on the first and the second law, and electrical efficiencies and the total system cost rate at different mixing ratios.

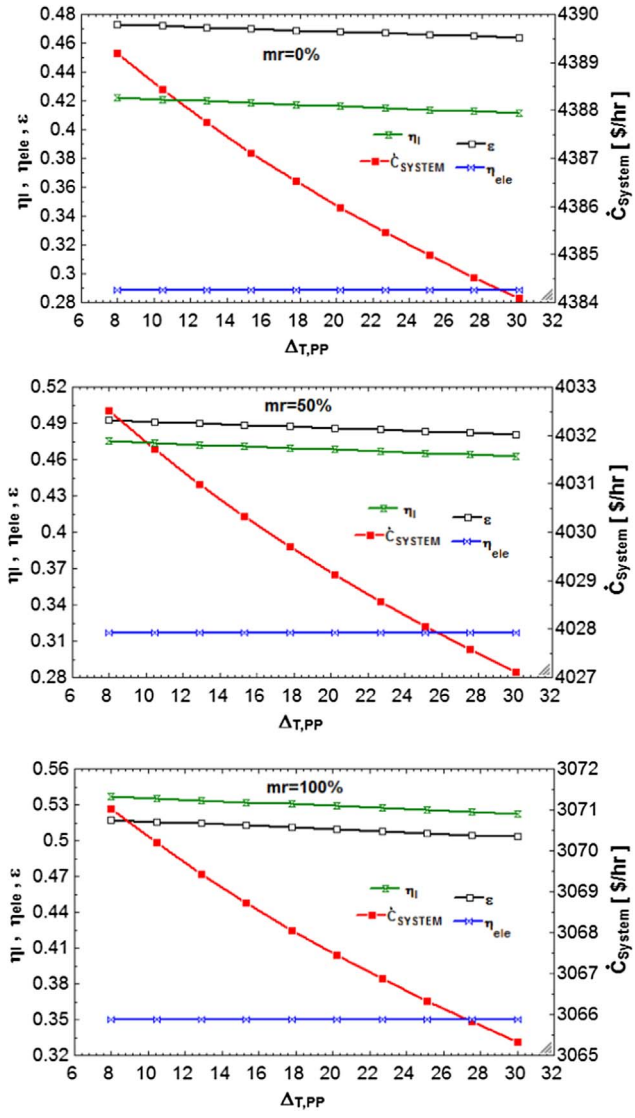

Fig. 11. The effect of pinch point temperature difference in evaporator on the first and the second law, and electrical efficiencies and the total system cost rate at different mixing ratios.

As it can be seen from this figure, increasing the percentage of biogas increases the total cost rate and the specific cost of the system and also reduces the second law efficiency. By decreasing the mixing ratio, the percentage of natural gas in the fuel mixture reduces and since the cost rate of biogas generation is higher than that of natural gas, consequently, the overall cost rate of the system increases.

There are several factors affecting the second law efficiency. Firstly, increasing the percentage of biogas in the fuel mixture which leads to a reduction in the exergy of the produced steam in the evaporator. Secondly, reduction in the exergy of the natural gas which is less than the increase in the exergy of the entering feedstock. From the other side, the exergy rate of the produced fertilizer from centrifuges is increased by the addition of biogas. All in all, increasing the percentage of natural gas in the fuel blend improves the system from thermodynamics view point. Also, Fig. 5 shows the effect of mixing ratio variation on the rate of the generated heat in the evaporator and the economizer.

The effect of variation in compressor pressure ratio on the performance and the cost criterion of the system is shown in Fig. 6, for three different mixing ratios of 0%, 50% and 100%, respectively. It can be observed that, the thermal efficiency, the electrical efficiency and the exergy efficiency of the cogeneration cycle increase by increasing the compressor pressure ratio. Although the compressor power demand is increased by increasing the compressor pressure ratio, the incoming air flow is reduced at this condition. As a result, the fuel consumption is decreased by reducing the incoming air flow rate which in turn leads to a decrease in the required input exergy to the system. On the contrary, by increasing the compressor pressure ratio, inlet water flow rate to the steam generator is reduced, and consequently causes to reduction in the heat load of the steam generator. All in all, the mentioned reasons raise the efficiencies.

As well as, increasing pressure ratio up to a certain amount, reduces the total cost rate of the system in all the mixing ratios. By increasing the pressure ratio, fuel mass flow rate decreases and leads to lower fuel cost rate, which in turn causes to a lower total system cost rate. But from a certain point onwards, for any mixing ratio, capital expenditures are increased. Hence, taking the sum of these changes results in a higher total system cost rate.

Fig. 7 indicates the effect of the air compressor isentropic efficiency on the considered performance and cost criterion of the system. As it can be seen from this figure, by increasing the isentropic efficiency of the air compressor, the thermal efficiency, the exergetic efficiency and the electrical efficiency of the cycle are increased, which is due to reduction in the input energy and exergy rates to the system. But higher

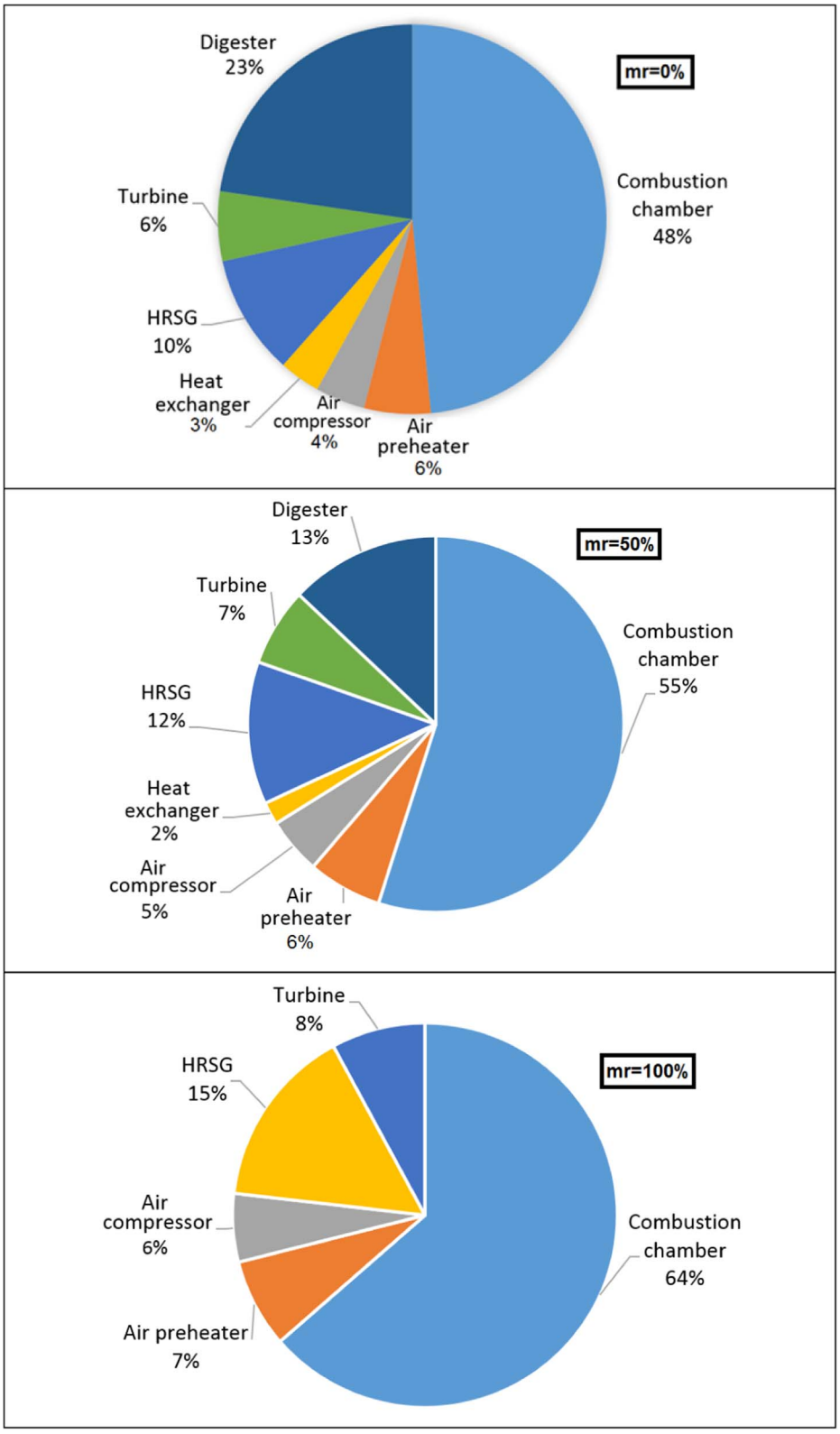

Fig. 12. Exergy destruction contribution of important components at different mixing ratios.

isentropic efficiency is not cost effective in terms of exergo-economic. By increasing the air compressor isentropic efficiency, compressor power consumption is reduced. Meanwhile considering constant net power output for the gas turbine cycle, compressor inlet air mass flow rate decreases, and consequently the input fuel flow rate is decreased. Therefore, fuel cost rate becomes lower, however with further increase in the isentropic efficiency of the air compressor, the capital costs of some components including compressor are drastically increased. Accordingly, by growing the isentropic efficiency of the air compressor, the total cost rate of the cogeneration system firstly decreases and then increases.

Fig. 8 demonstrates the effect of air pre-heater outlet temperature

on the considered performance and cost criterion of the system. By increasing the temperature of the incoming air into the combustion chamber, fuel flow rate is reduced which in turn reduces the required input energy and exergy into the combustion chamber that causes to increase in the electrical efficiency. Following that, flow rate and the exergy rate of the produced steam in the evaporator is reduced sharply. Hence, the sum of these changes makes the increases in the first and the second law efficiencies negligible in all the mixing ratios by increasing the air preheater outlet temperature.

Since the total cost rate is a function of fuel cost rate, therefore by increasing the inlet temperature of the combustion chamber in all the mixing ratios, the total cost rate of the system is decreased.

The effects of variation in gas turbine inlet temperature on the considered performance and cost criterion of the system are displayed in Fig. 9. Raising this parameter can increase the gas turbine output work and also the fuel mass flow rate. However, as the net power output is considered to be constant, the compressor inlet air mass flow rate is reduced and leads to reduction in the mass flow rate into the whole of the gas turbine cycle. Consequently, capital costs are reduced. However, more increase in gas turbine inlet temperature results in further increase in the fuel cost rate and also in the capital cost rates. These changes initially keep the total cost rate of the plant approximately constant, and then cause to increase in the total cost rate continuously for all the mixing ratios.

By increasing the fuel mass flow rate as a result of increase in the turbine inlet temperature, the energy and the exergy streams entering to the system are increased. On the other hand, the mass flow rate of the steam generated by the heat exchanger rises sharply which increases the energy and the exergy of the produced steam. Since the increase in the steam energy and exergy rates are greater than the increase in the input energy and exergy, the first and the second laws' efficiencies are improved. But electrical efficiency is reduced.

Fig. 10 represents the influence of variation in the turbine isentropic efficiency on the considered performance and cost criterion of the system. By increasing the turbine isentropic efficiency, the fuel mass flow rate is reduced. Reducing fuel mass flow rate causes to decrease in the energy and the exergy input to the system. Consequently, since the net power output is considered to be constant, electrical efficiency, the first, and the second law efficiencies increase. Reducing the natural gas and biogas flow rates leads to lower fuel cost rate and less capital costs. Also, the turbine purchase and maintenance cost is increased greatly by increasing the isentropic efficiency. Depending on the influence of the mentioned parameters, which one is dominant there would be an optimum point in which the total cost rate is minimum especially in pure natural gas.

Fig. 11 shows the effect of changing the pinch point temperature difference of the evaporator. By increasing the pinch point temperature difference of the evaporator, steam mass flow rate in HRSG and consequently the exergy rate of the generated steam is decreased, leading to a reduction in the first and the second law efficiencies. But, considering Eq. (34), it does not have effect on the electrical efficiency. Also, with more increasing the pinch point temperature difference of the evaporator, the capital cost rate of the heat exchangers and that of heat recovery steam generator are reduced in all the mixing ratios. Hence, by increasing the pinch point temperature difference of the evaporator, total system cost rate is reduced.

Also, it can be realized from Fig. 12 that in all the mixing ratios, the irreversibility of the combustion chamber is the highest value. After the combustion chamber, the anaerobic digester in the lower mixing ratios and the gas turbine in the higher mixing ratios have the second high irreversibilities.

6. Conclusions

In this study, the cogeneration of heat and power was investigated at different mixing ratios of biogas and natural gas from thermo-economic point of view, considering four objective functions of the first and the second law efficiencies, the electrical efficiency and the total system cost rate. The proposed cogeneration system comprises a 30 MW gas turbine cycle, a single-pressure HRSG to produce saturated steam at the pressure of 35 bar, an anaerobic digester, and a heat exchanger supplying the digester heat demand. First, the thermodynamic and thermo-economic models of the cogeneration cycle were developed and then the effect of various design parameters on the defined objective functions of the cycle was investigated.

Important conclusions are listed as follows:

- The most exergy destruction rate occurs in the combustion chamber, the anaerobic digester, the heat exchanger and the gas turbine at low mixing ratios, respectively, and it occurs in the combustion chamber, the steam generator and the gas turbine at high mixing ratios.
- By increasing natural gas contribution in the fuel mixing ratio, the first and the second law efficiencies are increased and also the total specific cost of the entire system is reduced.
- Increase in the isentropic efficiencies of the air compressor and the gas turbine, the gas turbine inlet temperature and the air compressor pressure ratio at all the mixing ratios cause to better system performance in terms of the first and the second laws' efficiencies. By increasing the gas turbine inlet temperature and the isentropic efficiency of gas turbine, the total cost rate of the cogeneration system increases and reduces respectively in all the mixing ratios. As well as the total cost rate of the cogeneration system rises with further increase in the air compressor pressure ratio and the isentropic efficiency of air compressor at all the mixing ratios. Increase in the pinch point temperature difference of evaporator in all mixing ratios causes to the lower first and second law efficiencies. Also, by increasing the pinch point temperature difference of the evaporator, and the outlet temperature of the preheater, the system cost rate are reduced in all mixing ratios. However, the first and the second law efficiencies do not change appreciably by rise in outlet temperature of the preheater.

According to the sensitivity analysis, carried out on the cycle, important parameters such as mixing ratio, air compressor isentropic efficiency, turbine isentropic efficiency, combustion chamber outlet temperature, outlet temperature of the preheater, the pinch point temperature difference of HRSG were identified as the main factors affecting the performance of the cycle. In a future work, the results of this study will be used for optimization of the cycle.

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